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EFFECT OF FIN PASSAGE LENGTH ON OPTIMIZATION OF CYLINDER HEAD COOLING FINS

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SUMMARY

The heat transfer performance of baffled cooling fins for use on cylinder heads of small, air-cooled, general-aviation aircraft engines was analyzed to determine the potential for improving cooling fin design. Current designs depend heavily on fuel cooling to augment air cooling during takeoff. Forthcoming pollution standards and the importance of fuel economy emphasize the need for improved air cooling. Flow baffles were assumed to be installed tightly against the end edges of the fins; this is an ideal baffle configuration for guiding all flow between the fins. With the baffles in place, a rectangular flow passage is formed between each set of two adjacent fins, the fin base surface, and the baffle. These rectangular passages extend around each side of the cylinder head, and the cooling air absorbs heat as it flows within them. For each flow passage length, the analysis is concerned with optimizing the fin spacing and thickness to achieve the best heat transfer for each fin width. Previous literature has been concerned mainly with maximizing the local fin conductance and has not considered the heating of the gas in the flow direction, which leads to higher wall temperatures at the exit of the fin passages. The present analysis treats the fins as part of a heat exchanger passage and specifically examines the effect of fin passage length. If the fins are close together, there is the advantage of a large surface area, but the airflow is restricted. This leads to high air temperatures at the rear of the cylinder head and consequently a high wall temperature at that location. Increasing the fin spacing provides more flow but may excessively reduce the surface area. Thus, there is an optimum fin spacing, and this spacing is investigated herein for each fin width as a function of the length of the fin passage in the flow direction. It was found that the optimum spacing increased as the flow passage length increased.

INTRODUCTION

According to the most recent census (1975) by the FAA (personal communication

with FAA headquarters, Washington, D. C.), there are 193 661 general-aviation airplanes in use in the United States. Almost all of these planes are powered by air-cooled reciprocating engines. The cooling fin design for these general-aviation engines emanated from the technology developed for large commercial and military aircraft engines in the 1930's and 40's. During that period, there was considerable work on the cooling of new, high-performance engines that entailed testing and analysis of fin and baffle geometries. Various dimensions and shapes of fins were tested and analyzed, and various fin materials were evaluated. Typical results from these studies are given in references 1 and 2. Reference 3 is a recent book that provides a good discussion of many aspects of engine cooling fin design.

Typically, in a small reciprocating aircraft engine about one-third of the energy from combustion is converted into useful mechanical work, about one-third is transmitted through the cylinder and cylinder head walls to the air-cooled fins and then into the cooling air, and the remaining one-third leaves in the exhaust gases. During takeoff there is an increased cooling load, and this has been met by fuel evaporation. As a result, during takeoff, about three-fourths of the fuel is burned, and the remaining one-fourth provides evaporative cooling and passes out through the exhaust. This method of cooling wastes fuel and is also an added source of air pollution. At cruise it is desirable to use a lean fuel mixture to conserve fuel, and this also reduces the heat removal by evaporative cooling. Consequently, air quality standards and fuel economy considerations have renewed interest in improving cooling fin design in order to reduce the amount of fuel required to cool general-aviation powerplants.

In the usual cylinder design, as shown in figure 1, a group of fins pass circumferentially around the cylinder barrel and around the cylinder head, thus forming a series of parallel plates that the cooling air passes between. Another group of fins extends along the top of the engine between and around the intake and exhaust ports. Because of the surface friction drag caused by the fins, the flow will tend to bypass the finned region if a less-restricted alternative flowpath is available. It may be particularly difficult to obtain adequate airflow into the corner regions where the fins meet the outside of the cylinder head wall; this is the most important heat transfer region and maintaining a high flow velocity is desired. To assure a substantial, controlled flow between the fin surfaces, it might be helpful to pay increased attention to the use of flow baffles, possibly making the baffles a fixed part of the engine structure rather than part of the nacelle. The baffling would be specified by the engine designer rather than left up to the airframe manufacturer. The analysis presented herein assumes that tightly fitting flow baffles are present.

With a flow baffle in contact with the fin edges, the flow geometry between two fins is a straight or curved rectangular duct; the duct walls are the adjacent surfaces of the two fins, the exterior surface of the cylinder head wall, and the baffle. The air flowing in each such passage is heated as it passes across the cylinder; thus, the air at the exit

of the finned region is at an elevated temperature and this can raise the wall temperature in this region. This temperature rise of the cooling air is a factor that has not been adequately treated in previous literature but is considered herein in optimizing the cooling fin geometry.

The temperature of the hot gases inside the cylinder can be as high as 2000° C (3630° F) (ref. 3). The aluminum-alloy cylinder head is at a much lower temperature, the recommended maximum temperature measured at the spark plug base being about 230° to 260° C (446° to 500° F) (refs. 4 and 5). The temperature on the inside of the wall can be as high as 315° to 345° C (600° to 653° F). Thus, the temperature difference between the outside of the cylinder head wall (at the base of the fins) and the cooling air is several times smaller than that from the combustion gas to the wall. Since the local heat load to the wall from the combustion gas must be dissipated from the outside of the wall, the heat transfer per unit temperature drop (conductance) from the outside of the wall (fin base) to the cooling air must be several times higher than that from the combustion gas to the inside of the wall. The large conductance from the exterior surface of the wall is achieved by using cooling fins. The literature typified by references 1 and 2 was concerned with optimization of the local fin conductance, that is, obtaining the maximum heat transfer per unit of exterior cylinder head wall surface and per degree of temperature between the wall and the cooling air. This optimum depends on fin thickness, spacing, width, etc. For example, if the number of fins per unit width across the surface is increased while the spacing between the fins is kept constant, more surface area will be exposed to the cooling air but the fin thickness will be reduced, thereby diminishing the ability of the fins to conduct heat away from the wall. An optimum fin thickness (and hence number of fins per unit of surface width) exists for each fin spacing.

The temperature variation of the air as it passes through the passages bounded by the fins and the baffles may be an additional variable in the optimization. For example, for a fixed fin thickness, increasing the spacing between the fins will reduce the fin surface area (which tends to raise the wall temperature), but there will be a compensating effect of increased airflow and hence reduced air temperature rise (which tends to lower the wall temperature). The coolant temperature effect is briefly considered in references 6 and 7 and is examined in more detail herein. A constraint in the optimization done herein is that there is a fixed pressure drop over the length of the fin passage. The results indicate that, to obtain the best performance, the spacing between the fins should be increased as the flow passage length along the fins is increased. Thus, the finned regions between the inlet and exhaust ports, which generally have short flowpaths, should have more closely spaced fins than the finned regions with long flowpaths extending around the circumference of the cylinder head. Fin spacing and channel length can also be used to help control the cooling air distribution; for example, providing a low flow resistance will allocate more cooling to a critical hot spot.

SYMBOLS

c_p	specific heat of cooling air
d_h	hydraulic diameter of passage between fins
f	friction factor for flow between fins
g	gravitational constant
H	external conductance of straight fins
H_f	external conductance of curved fins
h	convective heat transfer coefficient for cooling air
h_c	convective heat transfer coefficient for combustion gases to cylinder wall
k	thermal conductivity of cooling air
k_s	thermal conductivity of fin and wall material
l	length of passage between two fins
Pr	Prandtl number of cooling air
p	pressure
Q	heat flow per unit time
q	heat flow per unit area and time
Re	Reynolds number of cooling airflow
r	radial coordinate
s	spacing between fins
t	temperature
U	velocity of cooling air
w	width of cooling fins
x	coordinate along length of passage between fins
y	coordinate along fin width normal to fin base
z	coordinate parallel to fin base and normal to flow direction
δ	thickness of fins
δ_w	thickness of wall
θ	angular coordinate along curved passage
μ	viscosity of cooling air

ν kinematic viscosity of cooling air, μ/ρ

ρ density of cooling air

Subscripts:

c combustion gases

g cooling air

i inside surface of wall

o outside surface of wall

w wall

ANALYSIS

An aircraft engine cylinder head has an irregular shape. It is therefore necessary to use a few different groups of fins in order to cover the heat transfer surfaces around the top of the cylinder head and the exhaust port. There are usually a group of curved fins extending circumferentially around the head and a group of straight fins along the top of the head between the inlet and exhaust ports (fig. 1). In some designs the latter group of fins can extend obliquely between the inlet and exhaust ports so that enclosed rectangular passages are formed by the two port walls and the two adjacent fins; in this instance, flow baffles would not be required. In most instances, however, the fins extend outward from a wall surface. For these fins it is assumed that a baffle plate fits tightly against the fin tips so that the flow is confined in a passage between the fins. Hence, for this analysis the flow was always assumed to be guided in passages where each passage is bounded on two sides by parallel adjacent fin surfaces.

Figure 2 shows the models of the fins that are considered. The major group of fins extending circumferentially around the cylinder head is modeled in figure 2(a), and the straight finned regions between the inlet and exhaust ports are shown in figure 2(b). The derivation is carried out first for the straight fins. The straight fin geometry has the advantage that the fin length (l in fig. 2) does not change with fin width (w in fig. 2) so that the effects of length and width can be observed independently. (Note that "length" is along the flow direction and "width" is normal to the wall. This is the notation used in the engine literature.)

The flow in the rectangular passages between the fins was assumed to be turbulent, which is the usual mode of engine operation. During the calculations the Reynolds number was obtained, and the results included herein are only those where the Reynolds number was within the turbulent range.

Another assumption has to do with the transient nature of the heat transfer within a

reciprocating engine cylinder. The combustion gas temperature and the heat transfer coefficient are cyclic quantities that vary rapidly. The heat capacity of the wall will permit these variations to influence only a thin layer of the aluminum wall. Hence, for purposes of calculating heat dissipation, it is only necessary to use time-averaged values, and a steady-state analysis can be used. (Ref. 3, p. 63, states a ±2-percent error in calculated heat flow from this assumption.)

Straight Fins

From symmetry, a section of width $s + \delta$ can be considered to be composed of one fin and one space between fins, as shown in figure 3. The heat balances will be written first for a length dx along the fin channel; but before this is done, the nature of the heat flow into the cooling gas will be described.

The temperature of the combustion gases inside the cylinder t_c is high, from 1500° to 2000° C (2730° to 3630° F); the inside surface temperature of the cylinder head wall $t_{w,i}$ ranges from 260° to 345° C (500° to 650° F). As the cooling air passes along a fin passage and becomes heated, the wall temperature will also rise. However, the percentage change in $t_c - t_{w,i}$ cannot be large for any practical design because $t_{w,i}$ cannot become very high without exceeding the useful temperature range for the aluminum alloy. As a result, if the time-averaged heat transfer coefficient from the combustion gas to the wall h_c is assumed to be uniform over the wall surface, the quantity $h_c(t_c - t_{w,i})$ along the cylinder head wall does not vary over a large percentage range; thus, the q transferred through the wall is almost uniform everywhere. To reject this q through the external surface of the head wall then requires that for all locations the outside surface temperature of the wall (i.e., the temperature at the base of the fins) be greater than the local cooling gas temperature by a fixed amount. Thus, in regions where the cooling gas has become heated, there will be a corresponding rise in the outside surface temperature of the cylinder head wall. At the exit end of the fin passage the gas temperature is highest, and this will be the location of highest wall temperature. In this report the main result of the analysis is the inside wall temperature at the exit end of a channel between adjacent fins, as this is where the highest wall temperature will be reached. The fin optimization considered herein accounts for both the local fin performance and the behavior of the system as a whole, which involves the effect of the heating of the cooling gas as it passes between the fins.

The temperature of the fin base, which is the outside surface of the cylinder head wall, increases with x because the cooling air temperature rises in the flow direction. At each x the base temperature is assumed to be uniform; that is, it does not vary with the coordinate z in figure 3. The temperature gradient in the wall in the x-direction is much smaller than that through the wall thickness. Consequently, heat conduction is

neglected in the fin and wall in the flow direction, and the one-dimensional fin equation is applied locally at x. The heat loss from the edge of the fin at y = w is neglected, so that heat dissipation by conduction into the baffle plate is not taken into account. The fins on a cylinder head are usually somewhat tapered from base to tip, but because the taper is small, the analysis is simplified here by using straight fins having the average thickness of the tapered fins. Then from the fin equation (ref. 8) the heat flowing out within a fin from the base to the cooling airflow is

$$Q_{fin} = \sqrt{2hk_s\delta}\, dx[t_{w,o}(x) - t_g(x)]\tanh\left(\sqrt{\frac{2h}{k_s\delta}}\, w\right) \tag{1}$$

The heat flow from the wall surface between the fins to the cooling airflow is

$$Q_{space} = hs\, dx[t_{w,o}(x) - t_g(x)] \tag{2}$$

Combining equations (1) and (2) and dividing by the area of the fin base and the space give the heat flux from the finned surface as

$$q = \frac{Q_{fin} + Q_{space}}{(s+\delta)dx} = \frac{t_{w,o}(x) - t_g(x)}{s+\delta}\left[sh + \sqrt{2hk_s\delta}\tanh\left(\sqrt{\frac{2h}{k_s\delta}}\, w\right)\right] \tag{3}$$

For convenience, define an external conductance H (heat flux per unit temperature difference) of the fin system as

$$H = \frac{q}{t_{w,o}(x) - t_g(x)} = \frac{sh}{s+\delta}\left[1 + \frac{2}{s}\sqrt{\frac{k_s\delta}{2h}}\tanh\left(\sqrt{\frac{2h}{k_s\delta}}\, w\right)\right] \tag{4}$$

Using the overall temperature difference from the combustion gas to the local cooling air at x yields the local heat flux as

$$q(x) = \frac{t_c - t_g(x)}{\frac{1}{h_c} + \frac{\delta_w}{k_s} + \frac{1}{H}} \tag{5}$$

Using the temperature difference from the inside of the cylinder head wall to the cooling air gives

$$q(x) = \frac{t_{w,i}(x) - t_g(x)}{\frac{\delta_w}{k_s} + \frac{1}{H}} \tag{6}$$

Combining equations (5) and (6) to eliminate $q(x)$ yields

$$\frac{t_{w,i}(x) - t_g(x)}{t_c - t_g(x)} = \frac{\frac{\delta_w}{k_s} + \frac{1}{H}}{\frac{1}{h_c} + \frac{\delta_w}{k_s} + \frac{1}{H}} \tag{7}$$

To obtain the local wall temperature $t_{w,i}(x)$, the local air temperature $t_g(x)$ is needed. It is found from a heat balance on the flowing air by equating the heat gained by the air to the heat transmitted through the wall as given by equation (5),

$$ws U \rho c_p \frac{dt_g}{dx} dx = (s + \delta) dx\, q(x) = \frac{(s + \delta)[t_c - t_g(x)]dx}{\frac{1}{h_c} + \frac{\delta_w}{k_s} + \frac{1}{H}} \tag{8}$$

Equation (8) is a first order differential equation for t_g; it is integrated by separation of variables to yield

$$\frac{t_g(x) - t_c}{t_g(0) - t_c} = \exp\left(- \frac{s + \delta}{ws U \rho c_p} \frac{1}{\frac{1}{h_c} + \frac{\delta_w}{k_s} + \frac{1}{H}} x \right) \tag{9}$$

where the boundary condition has been imposed that $t_g = t_g(0)$ at $x = 0$.

To evaluate the previous relations, equations are needed for the mean air velocity U and the heat transfer coefficient on the external surfaces of the cylinder wall and fins. The condition that is fixed is that there is a given pressure drop available to force the flow between the fins. It is desired to optimize the fins such that for a fixed pressure drop the hottest temperature along the interior surface of the cylinder wall will be as low as possible. As discussed before, the region between adjacent fins forms a rectangular duct. Since in practice the flow is generally turbulent, the turbulent-channel-flow friction factor is used. The pressure drop through a fin channel is then given by

$$\Delta p = 4f \frac{\rho U^2}{2g} \frac{l}{d_h} \tag{10}$$

where the friction factor for fully developed flow can be approximated quite well by (ref. 9)

$$f = \frac{0.079}{Re^{1/4}} \tag{11}$$

The Reynolds number is $Re = Ud_h\rho/\mu$, where d_h is the hydraulic diameter of the rectangular passage.

$$d_h = \frac{2ws}{w + s} \tag{12}$$

By combining equations (10) to (12), the velocity is found from the pressure drop as

$$U = \left[\frac{\Delta p}{0.079 \left(1 + \frac{s}{w}\right)^{5/4} \frac{l}{s} \frac{\rho}{g} \left(\frac{\nu}{2s}\right)^{1/4}} \right]^{4/7} \tag{13}$$

The heat transfer coefficient is found from the relation for fully developed turbulent channel flow,

$$\frac{hd_h}{k} = 0.023 \left(\frac{Ud_h\rho}{\mu}\right)^{0.8} Pr^{0.4} \tag{14}$$

From equation (12), with some rearrangement, equation (14) becomes

$$h = 0.023\, k\, Pr^{0.4} \left(\frac{U\rho}{\mu}\right)^{0.8} \left(\frac{1 + \frac{s}{w}}{2s}\right)^{0.2} \tag{15}$$

A computer program was written that first obtained the flow velocity U from equation (13), where the pressure drop is assigned a specified value. The geometric

quantities - fin spacing s, fin width w, and fin length l in the flow direction - are also specified and, for obtaining the optimum dimensions, each of these quantities will be varied over a range while Δp remains fixed. The cooling air properties were taken at an average value along the fin channel; this required trial and error to some extent because the air temperature distribution was not found until later in the calculation. With the velocity U available, the heat transfer coefficient in the channel formed by the fins is obtained from equation (15). Then with the fin thermal conductivity and the fin thickness specified, the external conductance of the fin system is calculated from equation (4). With these quantities available the cooling air temperature along the channel can be found from equation (9). Then equation (7) can be solved for the interior cylinder wall temperature distribution. Of specific interest is the value of $x = l$ where the maximum wall temperature is obtained.

Curved Fins

A few calculations were done for the curved fin geometry shown in figure 2(a). The derivation in reference 1 shows that the external conductance in this case is slightly modified from equation (4) and has the form

$$H_f = \frac{sh}{s+\delta}\left[1 + \frac{2}{s}\sqrt{\frac{k_s\delta}{2h}}\left(1 + \frac{w}{2r_o}\right)\tanh\left(\sqrt{\frac{2h}{k_s\delta}}\,w\right)\right] \tag{16}$$

Because of the cylindrical geometry, equation (7) is modified to

$$\frac{t_{w,i}(x) - t_g(x)}{t_c - t_g(x)} = \frac{\frac{1}{k_s}\ln\frac{r_o}{r_i} + \frac{1}{r_o H_f}}{\frac{1}{r_i h_c} + \frac{1}{k_s}\ln\frac{r_o}{r_i} + \frac{1}{r_o H_f}} \tag{17}$$

Equating the heat gained by the cooling airflow to the heat transmitted through the cylinder wall gives

$$ws l\rho c_p dt_g = (s+\delta) r_o d\theta \frac{t_c - t_g(\theta)}{\frac{r_o}{r_i}\frac{1}{h_c} + \frac{r_o}{k_s}\ln\frac{r_o}{r_i} + \frac{1}{H_f}} \tag{18}$$

The variables are separated and equation (18) is integrated to yield the cooling air tem-

perature at angular position θ shown in figure 2(a)

$$\frac{t_g(\theta) - t_c}{t_g(0) - t_c} = \exp\left[- \frac{s + \delta}{wsU\rho c_p} \frac{1}{\frac{r_o}{r_i}\frac{1}{h_c} + \frac{r_o}{k_s} \ln \frac{r_o}{r_i} + \frac{1}{H_f}} r_o \theta \right] \tag{19}$$

where $\theta \approx \pi$ for the usual geometry, in which the flowpath between the fins extends about halfway around the cylinder.

To compute the velocity in the channel between the fins, the curved channel is replaced by an equivalent straight channel. The length of the straight channel is taken as the circumferential distance along the fins at the average fin height. Then, by letting l in equation (13) equal $\pi[r_o + (w/2)]$ for fins extending halfway around the cylinder head,

$$U = \left[\frac{\Delta p}{0.079\left(1 + \frac{s}{w}\right)^{5/4} \frac{\pi\left(r_o + \frac{w}{2}\right)}{s} \frac{\rho}{g}\left(\frac{\nu}{2s}\right)^{1/4}} \right]^{4/7} \tag{20}$$

The heat transfer coefficient is assumed to be uninfluenced by the duct curvature and hence can be calculated from equation (15).

RESULTS

The algebraic equations in the analysis can be evaluated to yield the cylinder wall temperature distribution after a number of geometric and heat transfer quantities are specified. Because of the number of quantities involved, it was not advantageous to combine variables into dimensionless groups. Attempting to nondimensionalize the equations tended to obscure the physical behavior and did not help to generalize the results. The approach taken herein was to select a set of physical conditions typical of an aircraft cylinder head and to calculate the heat transfer performance for various fin dimensions: thickness, spacing, width, and length. Then the interior surface temperature of the cylinder head wall at the exit of the fin channel was examined as a function of the geometric variables to determine what dimensions yielded the lowest wall temperature.

A wide range of numerical values can be chosen that are representative of engine performance. The heat transfer coefficient inside the cylinder not only varies with en-

gine type, but also cannot be specified to a high degree of certainty for a particular engine. In the calculations that follow, typical values have been used and the results should be considered as illustrative examples rather than as general design curves. The results show the important trends of how the different variables affect the optimum fin configuration and reveal the effect of fin passage length, which is the dimension under specific examination herein.

Since the heat flow originates from within the cylinder, the nature of this heat source must be specified for the calculations. The quantities required are the temperature of the combustion gases inside the cylinder t_c and the heat transfer coefficient from these gases to the internal cylinder wall surface h_c. These are rapidly time-varying cyclic quantities; but because of the heat capacity of the wall, the cyclic effects would be sensed by only a very thin layer of the aluminum cylinder head adjacent to the interior surface of the cylinder head wall. Hence, it is only necessary to use time-averaged values. Examples of t_c and h_c variations are given in figures 36 and 40 in chapter 3 of reference 3. From this information, time-averaged values characteristic of typical engine operation were selected for the present calculations as $t_c = 1650^{\circ}$ C (3000° F) and $h_c = 284$ W/(m^2)(K), or 50 Btu/(hr)(ft^2)($^{\circ}$F). The cylinder head wall was assumed to be 1.27 centimeters (0.5 in.) thick, and the fins and wall were assumed to be made of the same aluminum alloy with a thermal conductivity of 159 W/(m)(K), or 91.9 Btu/(hr)(ft)($^{\circ}$F). The cooling air properties were taken at approximately the average air temperature along the length of the fin channel. The incoming cooling air temperature was specified as 26.7° C (80° F). The pressure drop available to force air through the paths between the fins arises from the airplane velocity. Typical pressure drops (ref. 10, p. 8) used herein were in the range of 32.7 pascals per centimeter of passage length (4 in. H_2O/ft).

With these quantities specified, a fin geometry can be chosen and the temperature at the inside of the cylinder head wall can be calculated at the flow channel exit, which is the highest wall temperature. For straight fins the fin geometry consists of specifying the fin width, length, thickness, and spacing. For curved fins the cylinder radius must also be specified. The results for straight fins are given first; then for comparison some results are given for curved fins.

The interior wall temperature at the channel exit is plotted as a function of fin spacing with fin thickness held constant, or as a function of fin thickness with fin spacing held constant. This is done for various fin widths and lengths in the flow direction. From these curves it is evident which ranges of variables provide the best heat transfer performance and how wide a range of dimensions can be used without degrading the performance appreciably. The fin spacing and thickness are selected that give the minimum temperature for each fin width and length. A set of summary curves are then given that show the optimum combinations of thickness and spacing as a function of fin width and length.

Figure 4 shows results for a channel length of 30.5 centimeters (12 in.) and three fin widths, 1.3, 3.8, and 6.4 centimeters (0.5, 1.5, and 2.5 in.). Each curve is for a fixed fin thickness and shows the effect of varying the spacing between the fins. For example, consider figure 4(a), where four fin thicknesses are shown. For each thickness the curve passes through a minimum as spacing is varied. If the spacing is too small, there is insufficient airflow for a given pressure drop. If the spacing is too large, there are an insufficient number of fins along the surface, thus providing too little external surface area. There is a thickness that yields the lowest wall temperature; in the case of figure 4(a) this thickness is about 0.05 centimeter (0.02 in.) and the minimum temperature is at a spacing of about 0.42 centimeter (0.165 in.). Fins that are too thin have a poor fin efficiency because the outer portions of the fin are at temperatures significantly below the base surface temperature. Fins that are too thick occupy too much of the base surface area. Since these thick fins are nearly 100 percent efficient (i.e., their entire surface is essentially at the temperature of the base surface), an increase in fin thickness causes a reduction in heat transfer per unit wall area. Figures 4(b) and (c) show similar results for fin widths of 3.8 and 6.4 centimeters (1.5 and 2.5 in.). Figures 5 and 6 give similar results for fin passage lengths of 15.2 and 7.6 centimeters (6 and 3 in.). The spacing was not reduced below 0.15 centimeter (0.06 in.) because the flow Reynolds number in this range became too low for the flow to remain fully turbulent.

These results can also be crossplotted, as shown in figure 7. This figure shows the results of figure 4, and each curve is for a fixed spacing between the fins with the results plotted as a function of fin thickness. This figure is given to illustrate this type of plot. The most desirable fin thicknesses in these figures are 0.13 centimeter (0.05 in.) or less. From manufacturing considerations in the casting of aluminum cylinder heads, there is usually a minimum fin thickness somewhat larger than this that is practical, and we need to know the best fin spacing for this thickness. The curves in figures 4 to 6 are more useful for this purpose, and it is mostly the information from this type of curve that is presented herein.

Many of the calculated temperatures are higher than would be tolerated for the inside wall surface temperature of an aluminum cylinder head. The fins with a small width are clearly inadequate for the cooling required and might only be used if a much larger Δp were provided. It is evident that the cooling situation is a difficult one for the numerical values chosen (e.g., the average combustion gas temperature may be lower in many instances than that used herein) and that wide fins will be required. It is important to select dimensions that give performance near the optimum values. Since a wide range of input values could be selected, it is not the levels of the curves that are of most interest, but rather their shapes and relative positions.

The optimum fin dimensions are those that yield the minimum wall temperature; unfortunately, the optimum may not correspond to dimensions that are practical for

manufacturing purposes (i. e., the optimum fins may be too thin or too closely spaced to cast conveniently). In this instance the results of the type in figures 4 to 7 have the advantage that for other optimum thicknesses and spacings the curves show how far the wall temperature will be above the value obtained for optimum dimensions.

Even though they will not provide as much detailed information, such as off-optimum performance, it is of interest to look at the envelopes of the minimums for the various sets of curves in figures 4 to 7. Starting with figure 4, for each fin width the minimum was obtained for each curve of constant thickness, and the wall temperature at each minimum was then plotted as a function of thickness, with spacing as a parameter along the curve. This was done for the three fin widths in figure 4 as well as for two others, and the results are shown in figure 8. In a similar fashion the minimum points for the curves in figure 7 were plotted to yield the results in figure 9. The thickness-spacing combinations obtained at the minimums of the constant-thickness curves are different from the combinations obtained at the minimums of the constant-spacing curves. This can be realized by considering that the temperature is a function of spacing and thickness and that

$$dt = \left(\frac{\partial t}{\partial \delta}\right)_s d\delta + \left(\frac{\partial t}{\partial s}\right)_\delta ds$$

Along the constant-thickness curves there is a minimum where $(\partial t/\partial s)_\delta = 0$ and a different minimum where $(\partial t/\partial \delta)_s = 0$. The only common point is at the minimum for the entire envelope of curves, such as in figures 4 and 7; this is where both $(\partial t/\partial s)_\delta$ and $(\partial t/\partial \delta)_s$ are zero.

Figure 8 is for a fin passage length of 30.5 centimeters (12 in.). Similar curves were constructed for lengths of 22.9, 15.2, and 7.6 centimeters (9, 6, and 3 in.) and these are given in figures 10 to 12. The pressure drop per unit length was kept the same for all these curves, so they show only the effect of changing the fin passage length (i. e., the same flow velocity exists when fin spacing and width are kept constant). To show better the effect of passage length, the four curves for a fixed fin width but with four different lengths were plotted in one figure. Each of the three parts in figure 13 shows results for a different fin width, and the effect of fin length is quite evident.

A few results obtained for a larger pressure drop, 65.3 pascals per centimeter of passage length (8 in. H_2O/ft), are shown in figure 14 for a fin length of 15.2 centimeters (6 in.). This figure can be compared with figure 5, which has the same geometry but half the pressure drop. The main effects of the increased pressure drop are to improve the external heat transfer coefficient and to increase the coolant airflow velocity, thereby diminishing the heating of the gas flow as it passes through the fin passage. This causes a decrease in the wall temperature. The curves are shifted to somewhat smaller spacings, but the general nature of the curves has not changed appreciably even

though the pressure drop has been doubled.

Some calculations were also made for fins curved around a cylinder head. For these calculations the inside radius of the cylinder was taken as 7.0 centimeters (2.75 in.) and the wall thickness was retained at 1.3 centimeters (0.5 in.). The flow passage extends around one-half of the cylinder, as shown in figure 3(a). Results are given in figure 15 for pressure drops over the entire fin passage length of 995 and 1490 pascals (4 and 6 in. H_2O). These figures show the same trends in optimum dimensions as those obtained previously for the straight fins. Comparing figures 15(a) and (b) clearly shows the cooling increase that can be achieved by raising the available pressure drop.

DISCUSSION

The cylinder head wall temperature at the interior surface of the wall at the end of the fin passage, was examined as a function of the geometric variables. This is the highest wall temperature obtained and will provide a good measure of the cooling effectiveness of the fins.

The results in figure 4 for a fin length of 30.5 centimeters (12 in.) show that the fin thickness yielding the minimum wall temperature changes significantly with fin width. For a 1.3-centimeter (0.5-in.) width the minimum temperature is achieved at a thickness of about 0.051 centimeter (0.20 in.) with a spacing of 0.42 centimeter (0.165 in.). When the width is increased to 6.4 centimeters (2.5 in.), the optimum thickness increases to 0.13 centimeter (0.05 in.) and the spacing decreases to 0.28 centimeter (0.11 in.). The notable feature is that, for the 30.5-centimeter (12-in.) fin length and the larger fin widths, the region of minimum temperature is rather flat and broad, indicating an insensitivity to fin thickness or spacing. For example, for a width of 6.4 centimeters (2.5 in.) it does not make much difference which of the following combinations of thickness and spacing are used: $\delta = 0.10$ cm, $s = 0.25$ cm (0.04 and 0.10 in.); $\delta = 0.15$ cm, $s = 0.32$ cm (0.06 and 0.125 in.); $\delta = 0.20$ cm, $s = 0.38$ cm (0.08 and 0.15 in.).

When the fin length is reduced to 15.2 centimeters (6 in.) in figure 5, the minimums are at smaller spacings, and the minimum temperature is much more sensitive to the fin thickness. These trends are strengthened when the fin length is further reduced in figure 6. The fairly flat nature of some of the curves near their minimum suggests that a reasonable range of spacing can be used for a given fin thickness without much penalty in cooling performance. Increasing the spacing between the fins decreases the number of fins per unit area on the surface, thereby raising the wall temperature. However, it also increases the airflow, thereby lowering the temperature rise of the air; these two effects tend to compensate and lead to an insensitivity to fin spacing.

The calculations were made for fin spacings as small as 0.15 centimeter (0.06 in.). Smaller spacings than this are generally not used as they are difficult to manufacture by casting. The relations used in the analysis were not valid for smaller spacings because, for the range of pressure drops considered, the Reynolds number approaches that for laminar flow or for the transition region between laminar and turbulent flow.

The curves in figure 4, with the addition of results for fin widths of 2.5 and 5.1 centimeters (1 and 2 in.), were used to construct figure 8. This shows only the values of temperature and spacing where each curve of constant fin thickness passes through the minimum temperature. Each curve of constant fin width in figure 8 passes through a minimum, and this corresponds to the combination of thickness and spacing that provides the optimum performance. Figure 8 shows how this minimum shifts to smaller thicknesses and larger spacings as the fin width decreases. The wall temperature rises considerably as the fin width decreases and only the largest two fin widths provide temperatures that could be tolerated. These temperatures can be decreased by increasing the pressure drop, and some results for an increased Δp are shown in figure 14. Comparing figure 14 with figure 5 shows that the wall temperatures are decreased, but the curves have much the same relative behavior even though the Δp was doubled.

To consider a few specific values from figure 8, for a fin width of 3.8 centimeters (1.5 in.) the best performance is obtained at a fin thickness of 0.09 centimeter (0.035 in.) with a spacing of 0.31 centimeter (0.12 in.). If it should be necessary to increase the thickness (for the same fin width) to 0.15 centimeter (0.06 in.) for ease in casting, the spacing should be increased to 0.41 centimeter (0.16 in.). This design compromise is at a penalty of 4 degrees C (7 deg F) increase in wall temperature. A fin width of 6.4 centimeters (2.5 in.) has its best performance at a thickness of about 0.13 centimeter (0.05 in.) with a spacing of 0.28 centimeter (0.11 in.). If the fin thickness is increased to 0.15 centimeter (0.06 in.), the spacing should be increased to 0.32 centimeter (0.125 in.) to obtain the best performance at essentially no penalty in wall temperature change.

The results in figure 10 show that, if the fin channel length is reduced to 22.9 centimeters (9 in.) with the same pressure drop per unit length as in figure 8, the optimum thickness is shifted to smaller values. This shift continues as the length is further reduced to 15.2 and 7.6 centimeters (6 and 3 in.). These figures also clearly show the important effect of fin width.

To show better the effect of passage length, the results for a fixed fin width were consolidated in each part of figure 13. Taking figure 13(b) as an example, which is for a width of 3.8 centimeters (1.5 in.), the minimum temperature corresponds to a thickness of about 0.09 centimeter (0.035 in.) when the length is 30.5 centimeters (12 in.) but decreases to about 0.05 centimeter (0.02 in.) when the length decreases to 15.2 centimeters (6 in.). It is realized that these fins may be too thin to be practical for

casting. If a minimum practical thickness is chosen as 0.15 centimeter (0.06 in.), the optimum spacing would be about 0.22 centimeter (0.085 in.) for a length of 7.6 centimeters (3 in.) and would increase to almost 0.41 centimeter (0.16 in.) for a length of 30.5 centimeters (12 in.).

A cylinder head would typically have cooling flowpaths of various lengths through the finned regions. There could be a relatively short path through the finned region between the intake and exhaust ports and a relatively long path extending circumferentially around the head. Both paths have available about the same pressure drop to produce the flow. Consider an illustrative example. Let the short and long path lengths be 15.2 and 30.5 centimeters (6 and 12 in.) and the available pressure drop across the head be 995 pascals (4 in. H_2O). Let the fin width be 3.8 centimeters (1.5 in.) for both flowpaths and specify that it is not desirable that the fin thickness be less than 0.15 centimeter (0.06 in.). Then, from figures 4(b) and 14(b), the optimum spacing is 0.41 centimeter (0.16 in.) for the long length and 0.25 centimeter (0.10 in.) for the short length. This illustrates that, in general, the fin spacing should be increased in regions of long path length for adequate cooling.

In multiple-cylinder-head designs, the fin width is often cut down to save space in some regions, such as where two cylinders are adjacent to each other. This produces a region of short fins that are quite a bit thicker than they need to be for good efficiency in heat dissipation. (Typically these fins could be 1.3 centimeters (0.5 in.) wide and 0.15 centimeter (0.06 in.) thick.) As indicated by the calculations, such as figure 5, there could be considerable gain by cutting away a portion of the fin thickness, which would enlarge the spaces between them. For short fins of aluminum or other highly conductive material, it is not necessary to have much thickness for high fin efficiency, and a greater fin spacing benefits the cooling airflow.

CONCLUSIONS

The heat transfer performance of baffled fins for use in air-cooled, general-aviation aircraft engines was analyzed. The baffles were assumed to fit tightly against the fin edges so that they would channel the airflow between the fins. The base surface of the fins, the adjacent surfaces of two parallel fins, and the baffle at the fin edges form a rectangular flow passage with an aspect ratio determined by the fin width and spacing. It is recommended in practice that the fins be carefully baffled to take utmost advantage of the cooling air and pressure drop available. The analysis considered turbulent flow through such a passage and accounted for the heating of the cooling air along the passage, as well as the fin efficiency at each location along the passage length.

The criterion for good fin performance was to obtain a minimum interior wall surface temperature at the exit of the fin passage where the cooling air temperature is

highest. For a given fin length, width, and thickness, the spacing between the fins was varied to determine its effect on the exit wall temperature and in particular to obtain the spacing that provides the minimum temperature.

For a fixed fin width, the curves of exit wall temperature as a function of spacing were often rather flat in the vicinity of their minimums. Thus, there is a range of spacings that could be used without much change in performance. For the longer fin channel lengths the minimums for several of the curves of constant thickness were in the same temperature range. Thus, for a fixed fin width (the width would be dictated by the engine geometry), there is a range of spacings and corresponding thicknesses that will give about equal optimum cooling performance.

For a fixed fin width and a fixed pressure drop per unit of length along the fins, the length of the cooling passages between the fins had a substantial effect. The long lengths required thicker fins and larger fin spacings than the short lengths. This was a consequence of the heating of the cooling air along the passage. A larger spacing between the fins decreased the amount of fin surface per unit of wall area, but there was a net gain in cooling from the increase in airflow and the accompanying reduction in heating of the cooling air along the passage length. Thus, in a fin design for an engine, there should be various fin spacings, depending on the length of the finned region. The region between the inlet and exhaust ports is generally one of shorter length and higher available pressure drop per unit of length than the circumferential finned region around the head.

The results show that for fin widths less than about 5 centimeters (2 in.) the cooling capability of the fins diminishes significantly. Thus, it is beneficial to have at least 5 centimeters of fin width wherever space limitations permit.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, June 8, 1977,
505-05.

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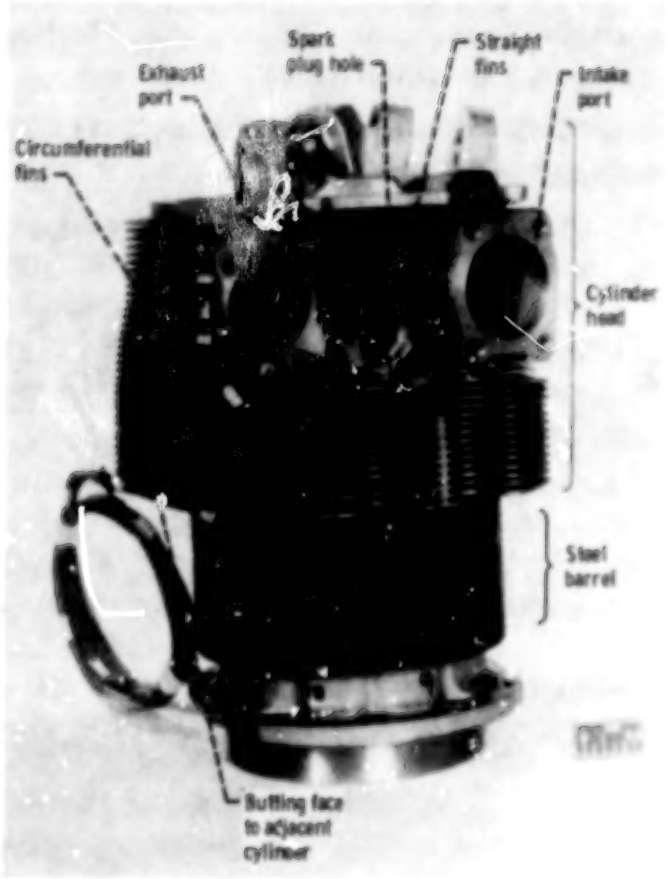
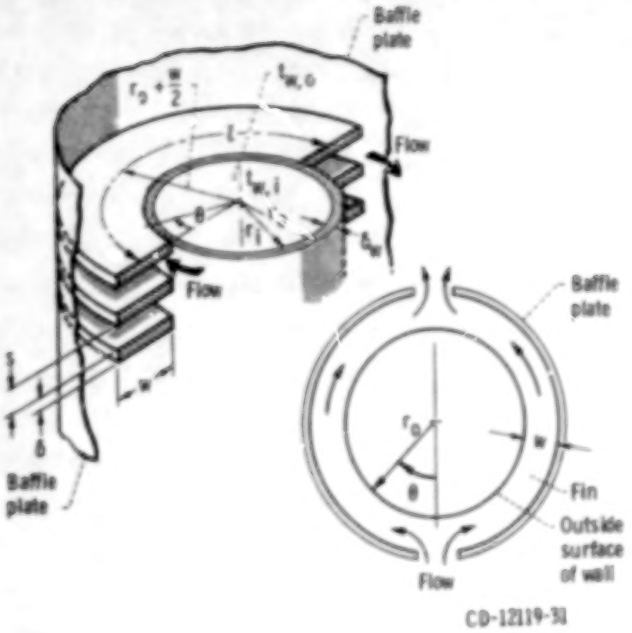
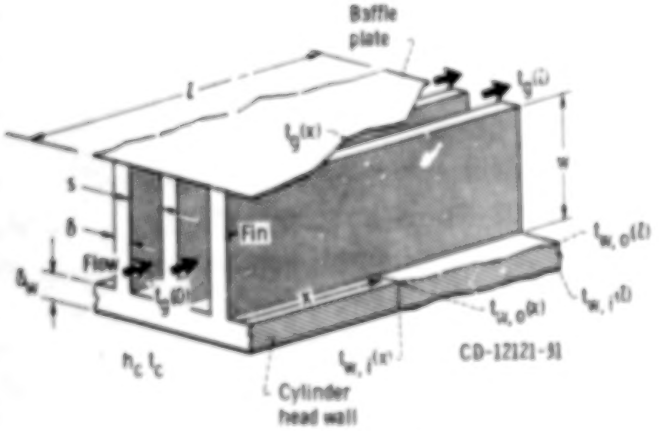


Figure 1. - Cast-aluminum cylinder head screwed onto machined-steel cylinder barrel.



(a) Circular fin geometry.



(b) Straight fin geometry.

Figure 2 - Geometry for analysis of fins.

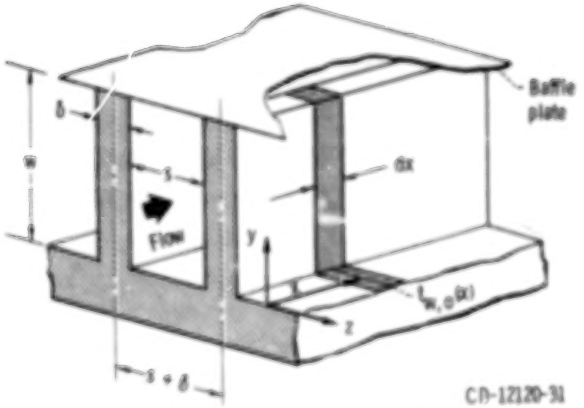
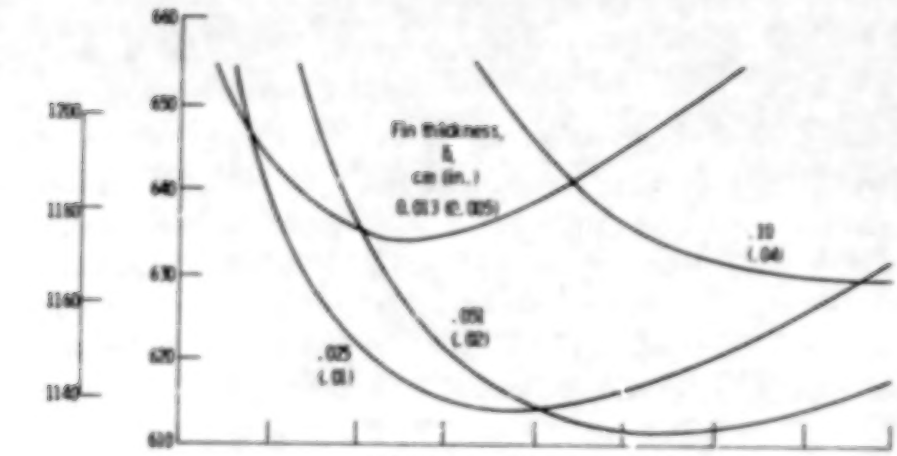
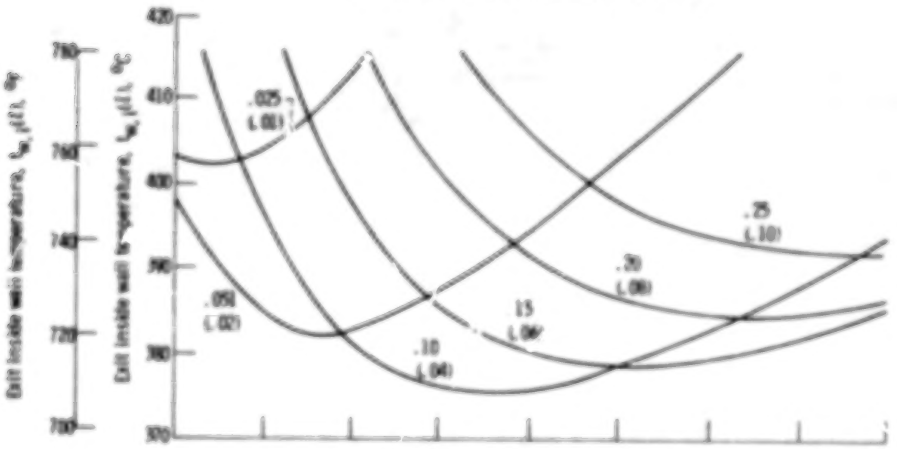


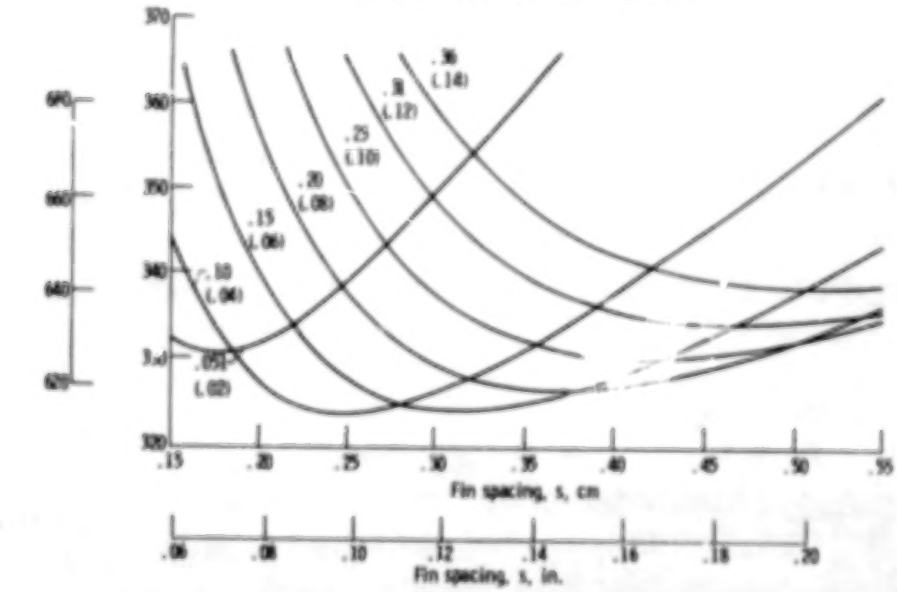
Figure 3. - Geometry for analysis of straight fins, showing section s + δ.



(a) Fin width, w, 1.3 centimeters (0.5 in.).



(b) Fin width, w, 3.8 centimeters (1.5 in.).



(c) Fin width, w, 6.4 centimeters (2.5 in.).

Figure 4. - Effect of fin spacing on exit wall temperature for straight fins 30.5 centimeters (12 in.) long. Pressure drop, Δp, 32.7 pascals per centimeter of length (4 in. H_2O/ft).

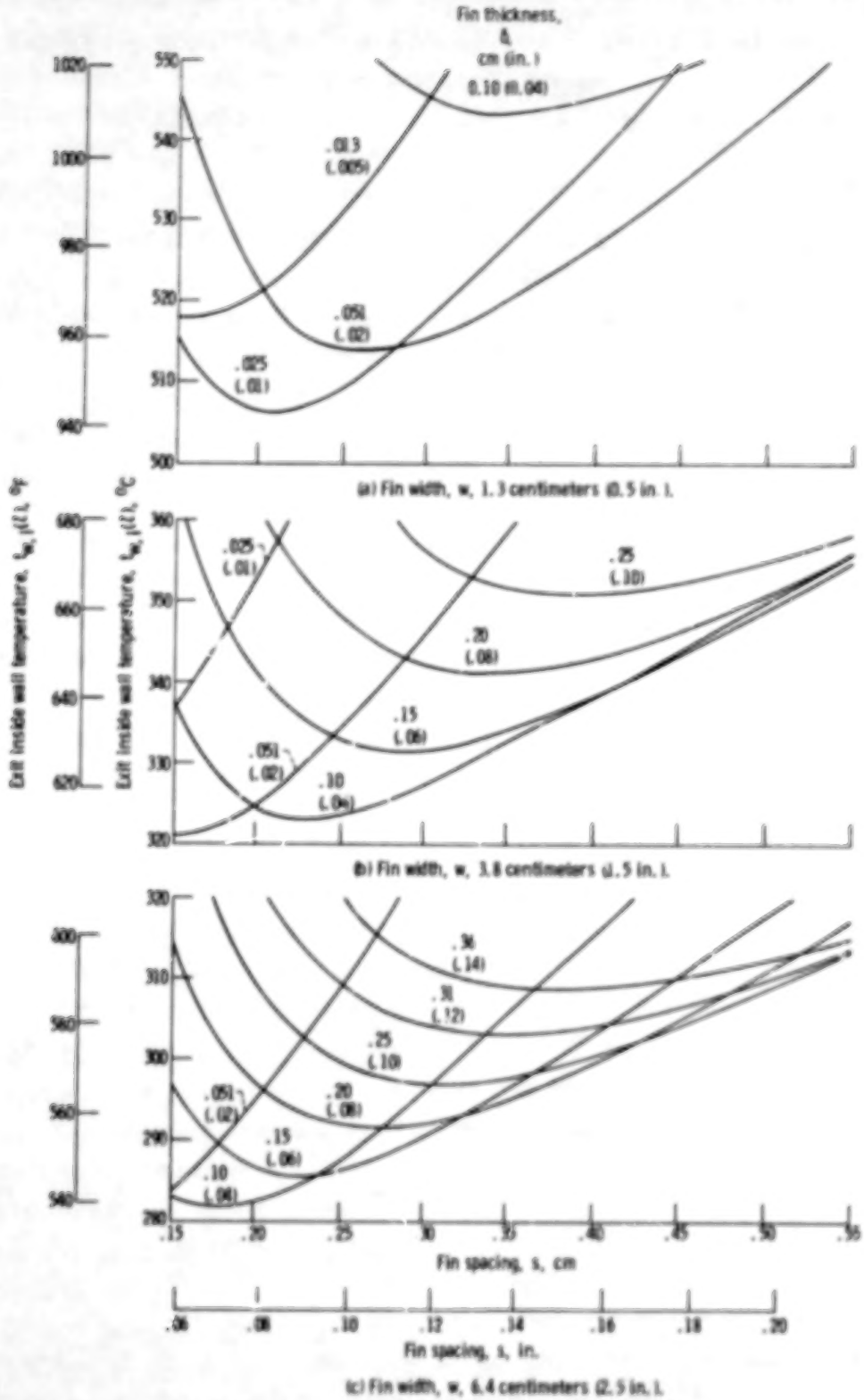


Figure 5. - Effect of fin spacing on exit wall temperature for straight fins 15.2 centimeters (6 in.) long. Pressure drop, Δp, 32.7 pascals per centimeter of length (4 in. H_2O/ft).

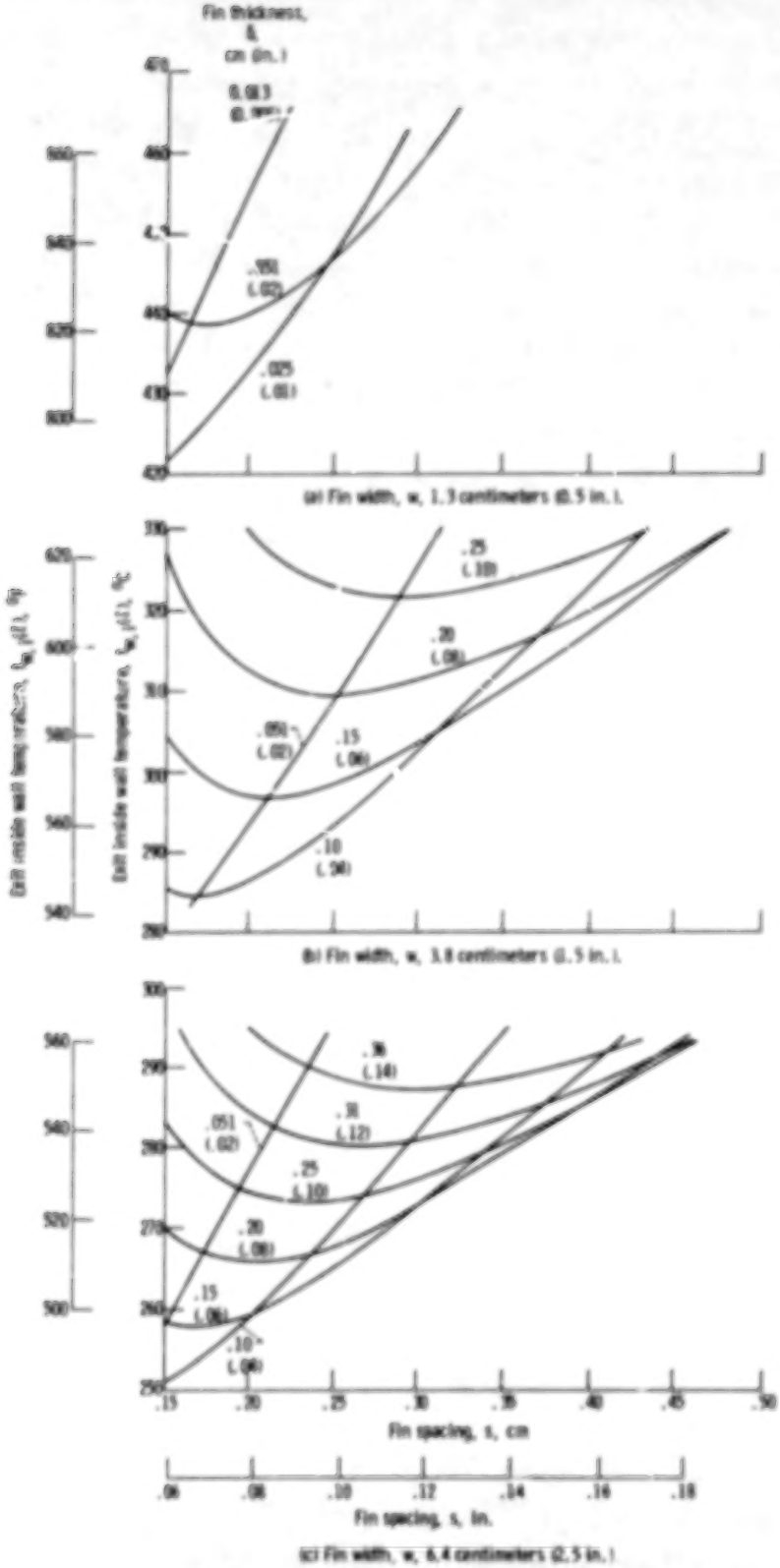


Figure 6. - Effect of fin spacing on exit wall temperature for straight fins 7.6 centimeters (3 in.) long. Pressure drop, Δp, 32.7 pascals per centimeter of length (1 in. H_2O/ft).

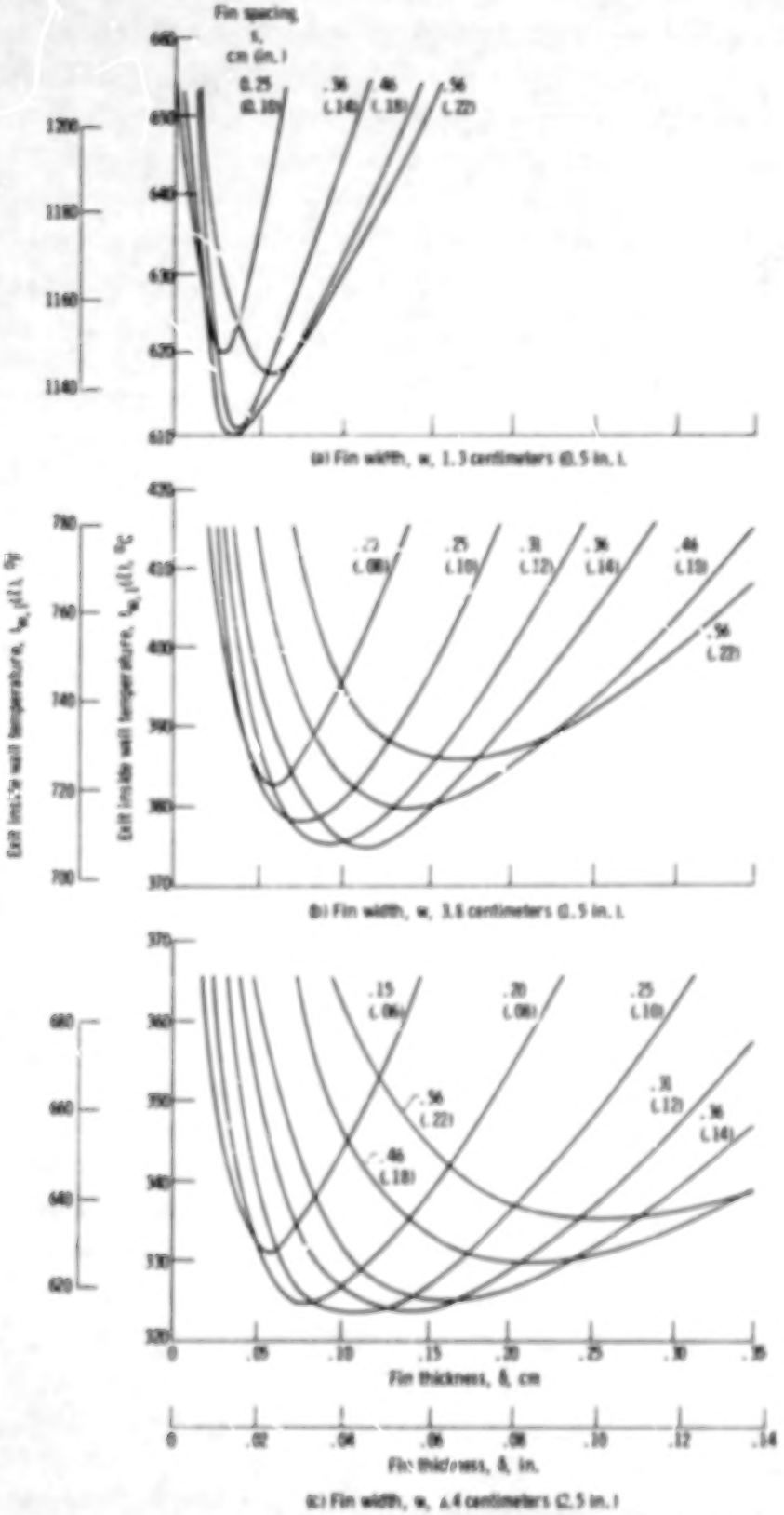


Figure 7. - Effect of fin thickness on exit wall temperature for straight fins 30.5 centimeters (12 in.) long. Pressure drop, Δp, 32.7 pascals per centimeter of length (4 in. H_2O/ft).

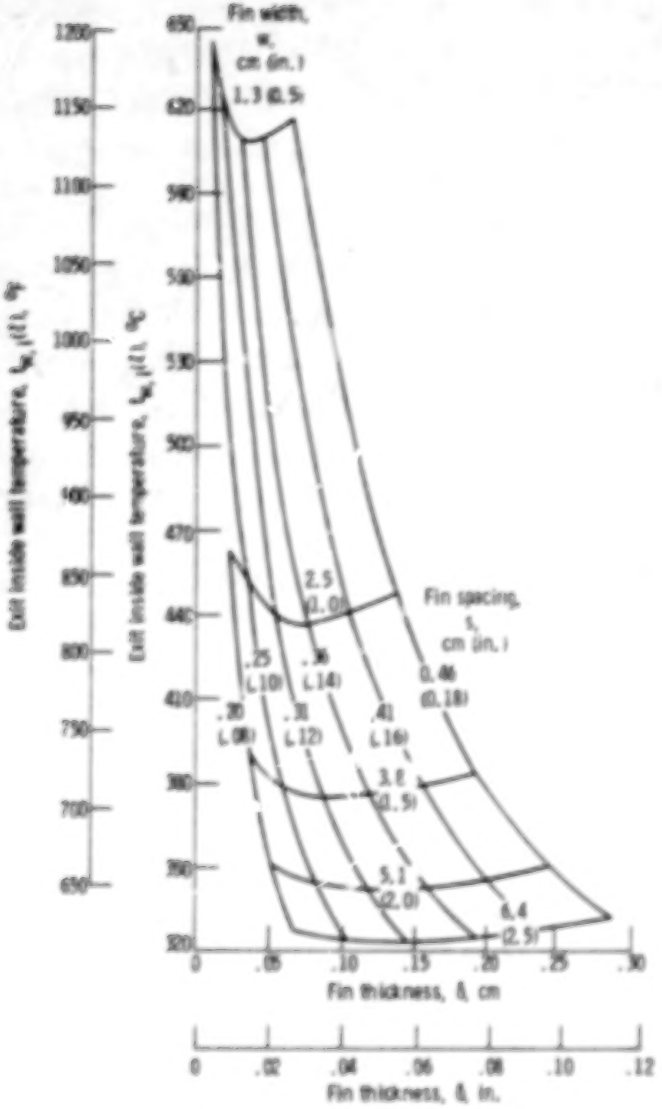


Figure 8. - Optimum combinations of fin thickness and spacing for straight 30.5-centimeter-long (12-in.-long) fins with specified thickness. Pressure drop, Δp, 32.7 pascals per centimeter of length (4 in. H_2O/ft).

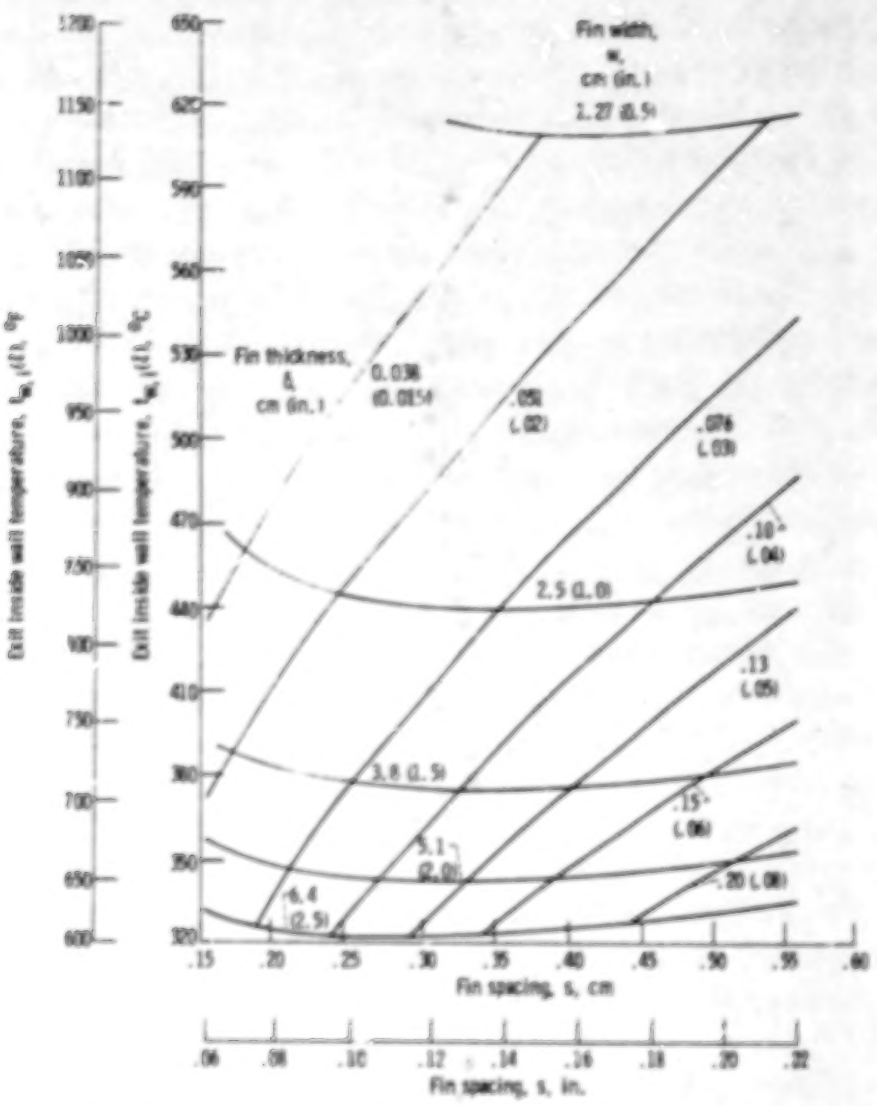


Figure 9. - Optimum combinations of fin thickness and spacing for straight 30.5-centimeter-long (12-in.-long) fins with specified spacing. Pressure drop, Δp, 32.7 pascals per centimeter of length (4 in. H_2O/ft).

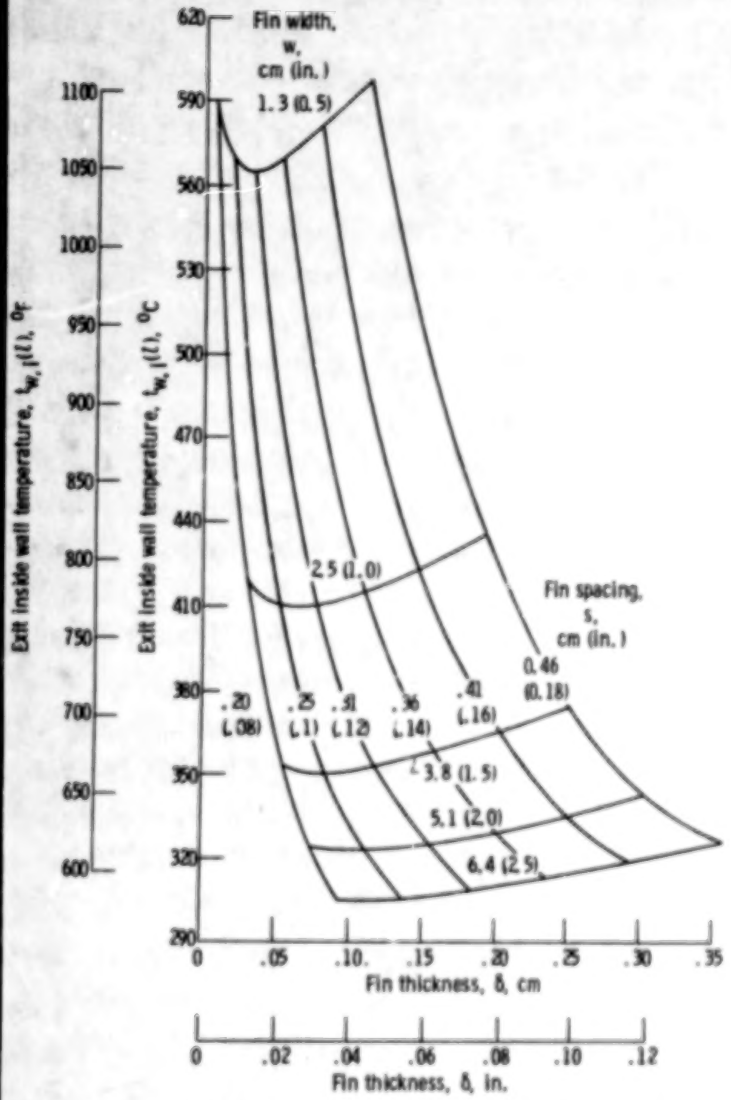


Figure 10. - Optimum combinations of fin thickness and spacing for straight 22.9-centimeter-long (9-in.-long) fins with specified thickness. Pressure drop, Δp, 32.7 pascals per centimeter of length (4 in. H_2O/ft).

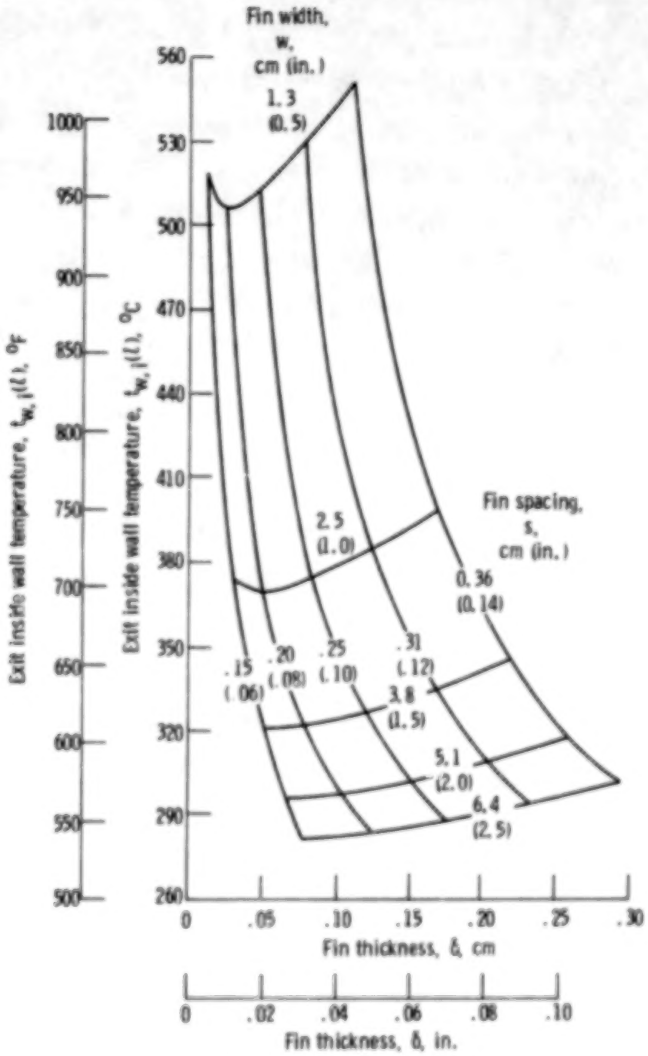


Figure 11. - Optimum combinations of fin thickness and spacing for straight 15.2-centimeter-long (6-in.-long) fins with specified thickness. Pressure drop, Δp, 22.7 pascals per centimeter of length (4 in. H_2O/ft).

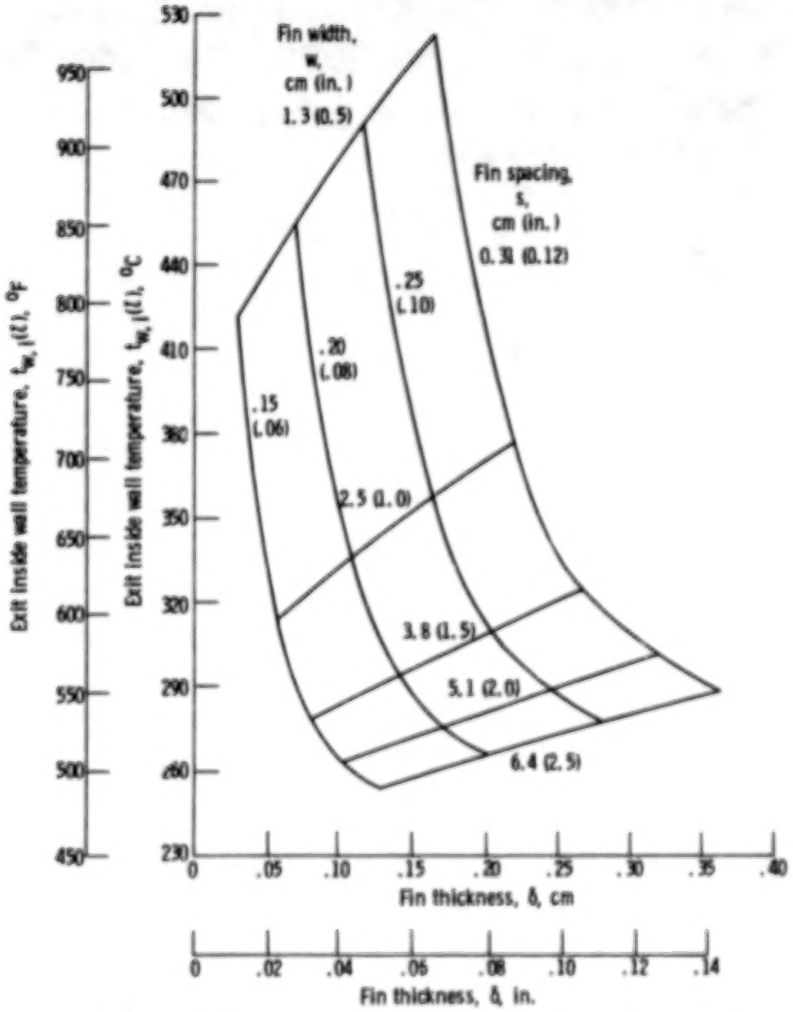


Figure 12. - Optimum combination of fin thickness and spacing for straight 7.6-centimeter-long (3-in.-long) fins with specified thickness. Pressure drop, Δp, 32.7 pascals per centimeter of length (4 in. H_2O/ft).

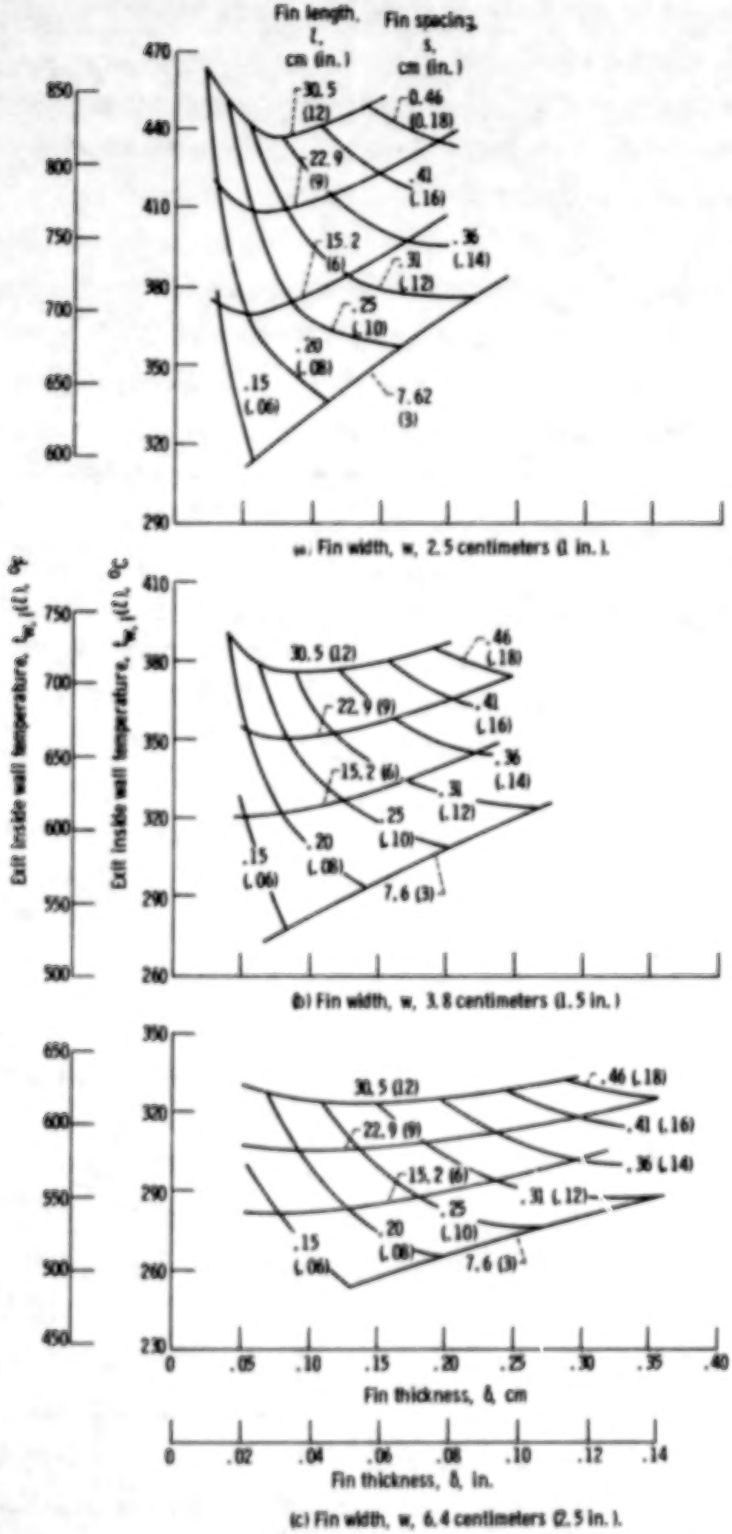

Fin length, l, cm (in.)
Fin spacing, s, cm (in.)
Exit inside wall temperature, $t_{w,i}(L)$, °F
Exit inside wall temperature, $t_{w,i}(L)$, °C
(a) Fin width, w, 2.5 centimeters (1 in.).
(b) Fin width, w, 3.8 centimeters (1.5 in.)
(c) Fin width, w, 6.4 centimeters (2.5 in.).
Fin thickness, δ, cm
Fin thickness, δ, in.

Figure 13. - Effect of fin length on optimum combinations of fin thickness and spacing for straight fins with specified thickness. Pressure drop, Δp, 32.7 pascals per centimeter of length (4 in. H_2O/ft).

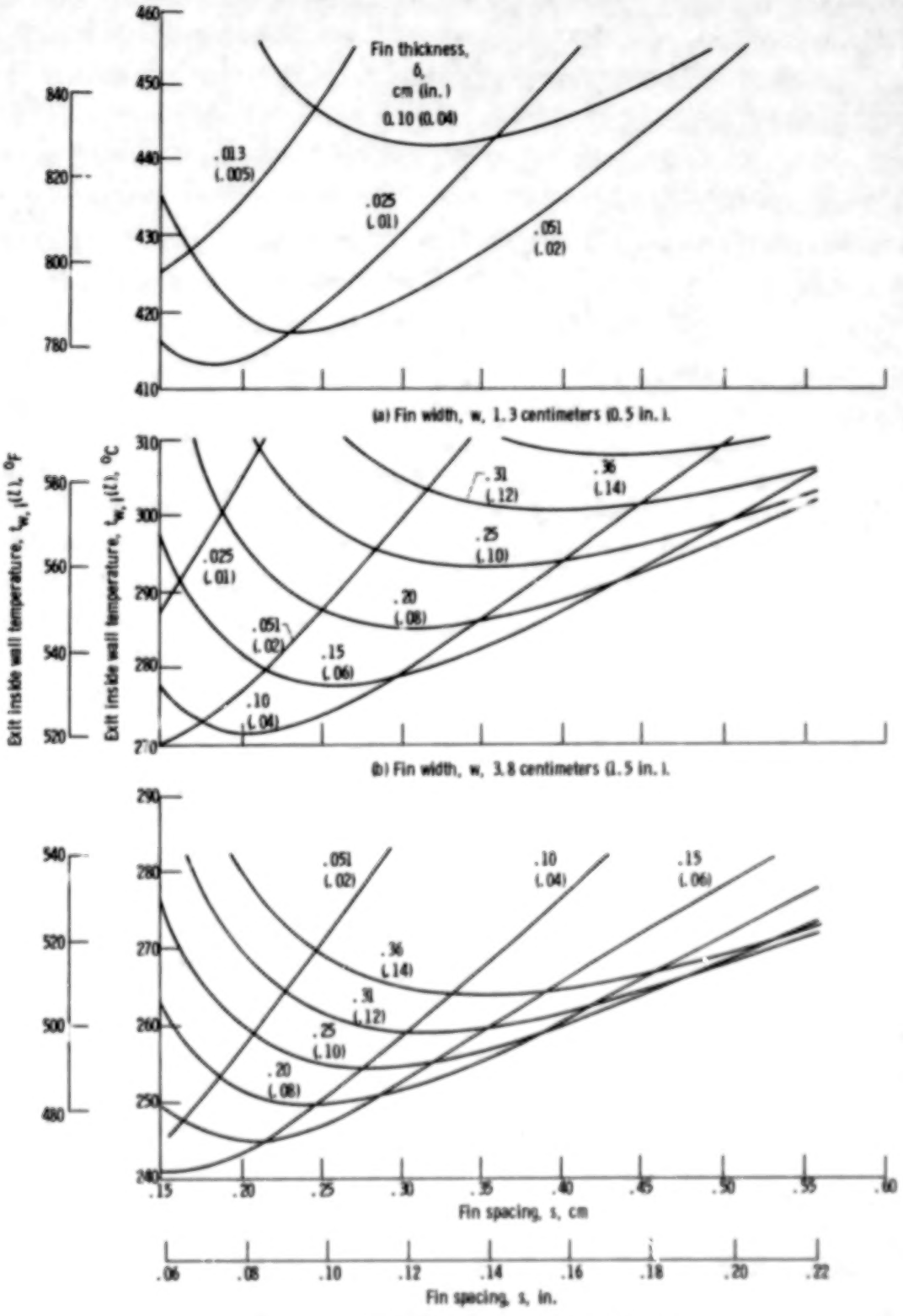
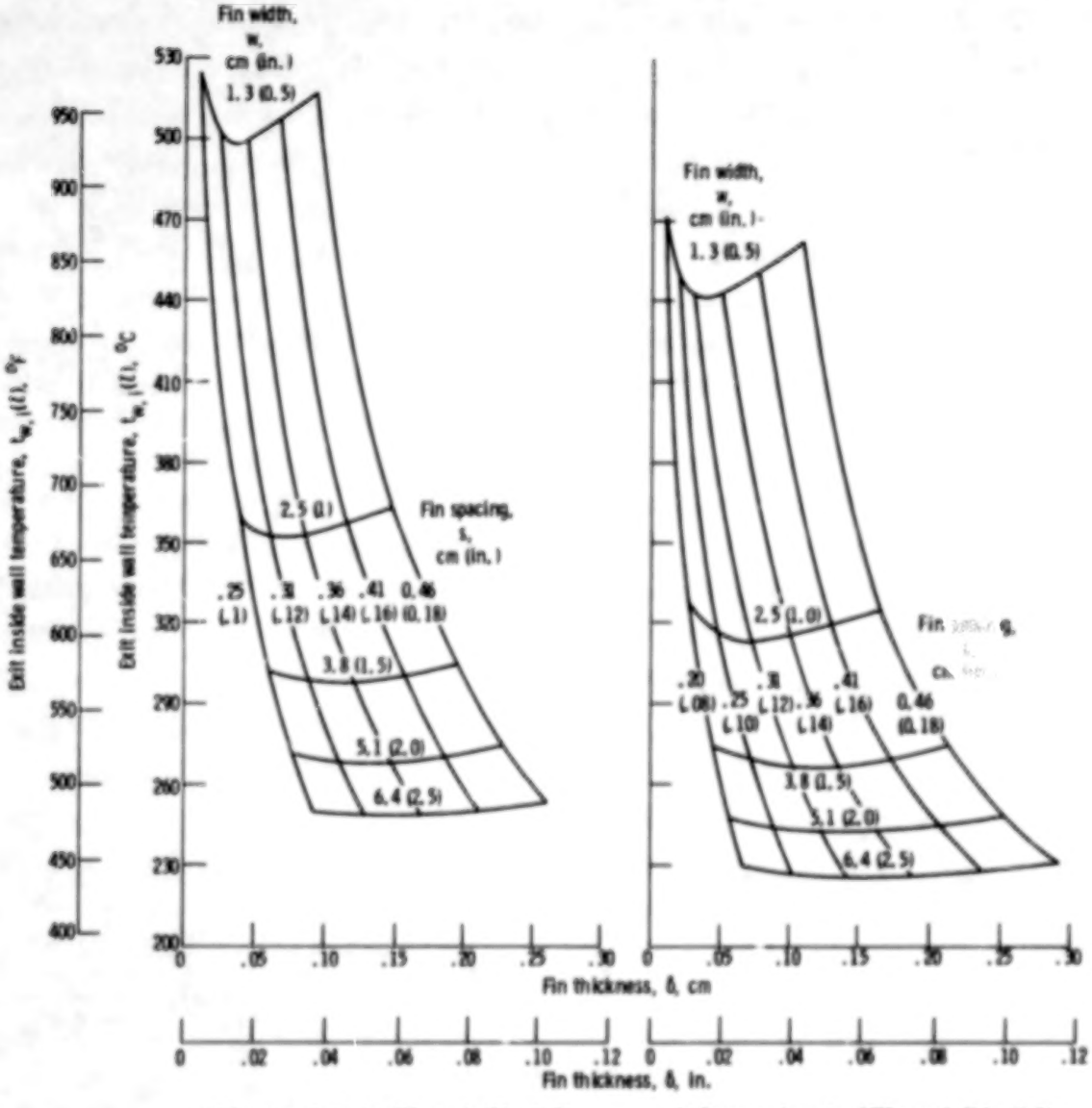


Figure 14. - Effect of fin spacing on exit wall temperature for straight 15.2-centimeter-long (6-in.-long) fins. Pressure drop, Δp, 65.3 pascals per centimeter of length (6 in. H_2O/ft).



(a) Pressure drop, Δp, 995 pascals (4 in. H_2O).

(b) Pressure drop, Δp, 1490 pascals (6 in. H_2O).

Figure 15. - Optimum combination of fin thickness and spacing for curved fins with specified thickness.

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15. Supplementary Notes

16. Abstract

The heat transfer performance of baffled cooling fins on cylinder heads of small, air-cooled, general-aviation aircraft engines was analyzed to determine the potential for improving cooling fin design. Flow baffles were assumed to be installed tightly against the fin end edges, an ideal baffle configuration for guiding all flow between the fins. A rectangular flow passage is thereby formed between each set of two adjacent fins, the fin base surface, and the baffle. These passages extend around each side of the cylinder head, and the cooling air absorbs heat as it flows within them. For each flow passage length, the analysis was concerned with optimizing fin spacing and thickness to achieve the best heat transfer for each fin width. Previous literature has been concerned mainly with maximizing the local fin conductance and has not considered the heating of the gas in the flow direction, which leads to higher wall temperatures at the fin passage exits. If the fins are close together, there is a large surface area, but the airflow is restricted. This leads to high air temperatures at the rear of the cylinder head and consequently a high wall temperature at that location. Increasing the fin spacing provides more flow but may excessively reduce the surface area and lead to high wall temperatures. Thus, there is an optimum fin spacing, which is investigated herein for each fin width as a function of fin passage length in the flow direction. The optimum spacing increased as the flow passage length increased.

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